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# VIBRATION FATIGUE STUDY OF ELASTOMERIC SHEAR MOUNTS

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# VIBRATION FATIGUE STUDY OF ELASTOMERIC SHEAR MOUNTS

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## SYMBOLS

| | |
|---|---|
| a | Acceleration (in. /sec$^2$) |
| c | Damping constant (lb/in. per sec) |
| $c_c$ | Critical damping coefficient (lb/in. per sec) |
| cps | Cycles per second |
| DA | Double amplitude (in.) |
| F | Force (lb) |
| $F_o$ | Stead force (lb) |
| f | Frequency (Hz or cps) |
| $f_n$ | Natural frequency (Hz or cps) ( ≈ resonant frequency) |
| g | Acceleration of gravity (386 in. /sec$^2$) |
| Hz | Hertz (equivalent to cps) |
| k | Spring force (lb/in.) |
| m | Mass (lb/in. per sec$^2$) |
| Q | Peak transmissibility |
| t | Time (sec) |
| W | Weight (lb) |
| X | Displacement of exciting force (in.) |
| $X_o$ | Zero frequency deflection of system (in.) |
| x | Displacement (in.) |
| $\dot{x}$ | Velocity or first derivative of displacement with respect to time (in. /sec) |

$\ddot{x}$ Acceleration or second derivative of displacement with respect to time (in. $\sec^2$)

$\zeta$ Damping factor

$\phi$ Phase angle

$\omega$ Angular frequency (rad/sec)

$\omega_n$ Natural angular frequency (rad/sec)

## OBJECTIVES

1. To develop a simple yet reliable method of testing and comparing the fatigue characteristics of various types of elastomer as they function in sandwich-type shear mounts used in isolation systems.

2. To define failure in elastomeric shear mounts and measure the parameters involved.

3. To test samples of typical shear mounts to obtain statistical data.

## SUMMARY

A method of measuring the fatigue life of elastomeric shear mounts was developed. It included vibration testing at resonance to failure at different inputs, and plotting the time to failure vs input displacement for each set of mounts tested.

Twelve pairs of mounts composed of butadiene styrene elastomer were subjected to static load deflection and vibration testing. The nominal static spring rate of one mount was 470 pounds/in. The load deflections yielded results within 7% of this value.

During initial vibration the transmissibility (Q) was found to vary between 3.1 and 4.4 and the resonant frequency ($f_n$) between 8.4 and 9.2 Hz. Eleven pairs of mounts were fatigue-vibrated at double amplitudes (DA) of 1/8", 0.2", 1/4" and 1/2". The durations to failure were approximately four hours for 1/8 inch and five minutes for 1/2 inch.

What actually constitutes failure in a shear mount has been a matter of concern for some time. After investigating changes, if any, in frequency, transmissibility, etc, failure was defined as a visual tear or separation of the mount.

Upon completion of the fatigue vibration testing, five pairs of mounts were again subjected to load deflection testing. The loss of static spring rate amounted to less than 15% for all five deflected up to 3 inches, and in most cases it was less than 10%.

## CONCLUSIONS

The method of testing fatigue, i.e., input displacement vs time to failure at resonance, proved satisfactory. It should be expected that once a curve has been generated for a specific composition (butadiene styrene, silicone, etc) all mounts composed of this same material should fall within the scope of that curve.

Heat appears to be a factor in the deterioration and failure of butadiene styrene mounts. A rapid heat buildup was observed, and the final temperature reached was relatively high.

Butadiene styrene mounts exhibit typical patterns of failure under fatigue loading, which means longer life at lower stress levels, and reduced life at higher levels. At 1/8 in. input DA and resonance, the average time to failure was just over 4 hours, while at 1/2 in. input DA and resonance, it was only 4.8 minutes. Based on information supplied by the manufacturer it had been estimated that the mounts would have a life of approximately 30 minutes at 1/2 in. input. Thus, the time period of 4.8 minutes was unexpectedly short.

The definition of failure as a visual tear or separation of the mount was the only logical conclusion that could be drawn from the available information. At the actual time of failure no significant change in either the resonant frequency or transmissibility was evident. Also, the curves generated during the static load deflection tests, performed on five pairs of mounts after vibration failure, corresponded very closely to those obtained before vibration.

The relatively small losses in spring rates, together with the stability of the frequency and transmissibility at failure, would indicate little change in the mount's resilience. It would appear that even a suspended item with torn mounts could still be protected from the environment. However, a physical sign of deterioration must be considered as a criterion for failure.

# RECOMMENDATIONS

This report should be considered only a first step in the fatigue study of elastomeric shear mounts. More extensive testing is needed and should be conducted with different types of mounts. Mounts with different characteristics and configurations as well as material compositions should be included, and comparisons drawn among them. Temperature extremes, especially high ones, will have to be taken into consideration. Indeed, the relationship between temperature and strain will be of prime importance. Mounts produced from silicone compounds, e.g., should stand up better than others under high temperatures and large deflections, and this should be investigated.

The investigation of fatigue due to shock (drop tests) and fatigue due to shock in conjunction with vibration should also be undertaken. Low temperatures will be most important, due to stiffening of the mounts, and a method of accurately measuring the temperature in shock as well as vibration should be developed.

Some difficulty was encountered in detecting tears or separations in the elastomer, which were defined as failure. Consideration therefore should be given to modifying the vibration test fixture so as to increase the visual observation capabilities of the mounts during vibration. This might be accomplished by cutting a hole in each of the end plates, or by replacing them with transparent plexiglass. Due to strength considerations, the former seems more feasible.

## INTRODUCTION

Any properly designed system employing sandwich-type elastomeric shear mounts will have no difficulty in passing standard static and dynamic loading tests. However, such tests generally will give no indication of the fatigue life of the mounts or how much of that life is removed by individual testing. The only definite information obtained would be to guarantee the fatigue life to be greater than the limits of the standard vibration and shock tests.

Pure fatigue testing of shear mounts has been neglected in the past. Very little thought, if any, had been given to a method of either accomplishing or measuring this type of failure. As a consequence it was decided to perform tests in both vibration and shock to determine the fatigue life, under varying load conditions, of different types of shear mounts. A number of different methods of accomplishing this were available, and they all had to be considered for reliability, ease of performance, cost, and adaptability to existing equipment. The methods considered were fatigue studies of mounts in either an actual shipping container or in a test fixture. Also, it had to be determined whether to use sweep frequency vibration or resonant point vibration, and whether to use shock studies in conjunction with vibration or separately.

Any method employing a shipping container was rejected because of cost, handling difficulties, and nonadaptability to the present equipment due to the size of the container. Also, the fact that a container is in itself an elastic member and will not transmit all of the imposed force through to the shear mounts could lead to unreliable and often nonrepeatable results.

Sweep frequency vibration according to any of the standard procedures (PA-PD-2522, MIL-STD-810B, etc) was not employed due to the durations involved and inappropriateness of higher frequencies. The time to failure would have been extremely long even with large inputs because of relatively small displacements of the mounts beyond the resonant point. (At higher frequency levels the mounts are practically stationary).

Although shock testing in conjunction with vibration might yield interesting and useful results, it was not performed in this sequence of tests. Later studies are planned to obtain shock data both separately and in connection with vibration.

In dealing with vibratory inputs as opposed to shock, more difficulty is encountered in attempting to simulate actual conditions. Vibration encountered in actual use varies continuously as a function of time (is random); and the inputs (displacements) so induced in the system are at the frequencies associated with the forcing function. Such random vibration is very difficult and costly to simulate in a laboratory. However, little damage results to a shear mount when vibrated beyond its resonant frequency because of the small displacements involved. Therefore, a varying vibration, especially at higher frequencies, does not appear necessary.

The method chosen consequently incorporated two mounts in a rigid test fixture and continuous vibration to failure at the resonant frequency of the system. The time durations so obtained should be reasonable in length and the existing test fixture was expected to yield useful information.

Information obtained about the fatigue characteristics of elastomeric shear mounts also will be of great significance in the future design of mounting systems and in the evaluation of newly developed mounts. A relationship between frequency, amplitude, damping, etc and fatigue life for specific materials may be used to estimate a time to failure for different applications.

## BACKGROUND THEORY

Both private industry and the military sector today produce numerous delicate items that require sophisticated packaging designs to protect them from shock and vibration damage. In use for some time now, isolation systems employing sandwich-type elastomeric shear mounts have proven to be more than adequate in meeting the task when utilized properly. Their primary uses have been in the field of missile containers, but wider applications are being investigated at present.

However, very little is known about the fatigue characteristics of shear mounts. Of prime consideration is the time to failure under varying conditions and for varying causes. In the past, mounts were simply tested to determine their basic characteristics (static and dynamic spring rates, natural frequencies at various load levels, etc) and their ability to protect an item in different environments. There

was no desire for information concerning the ultimate fatigue life or the factors that influence it. It was the objective of this program to shed some light on this area.

In order to simplify our approach and still maintain validity, certain restrictions had to be placed upon the test setup. The mounts and the suspended mass were allowed to move in only one plane. This reduced the motion to a single-degree-of-freedom system with its associated equations of motion. A view of this equipment is shown in Figures 1a and 1b.

All equations of motion are applications of Newton's Second Law of Motion, $F = ma$, when viewed within a Newtonian frame of reference. If the force F is replaced by the spring factor k, multiplied by the displacement x from Hooke's Law, we obtain

$$-kx = ma \qquad (1)$$

where m is the mass of the item and a the acceleration. The minus sign appears because the spring force of the mounts will oppose a forcing function.

Rearranging the equation and substituting $\ddot{x}$ for a ($\ddot{x}$ indicates the second derivative of the displacement with respect to time or acceleration; $\dot{x}$ would be the first derivative, or velocity) we obtain

$$\ddot{x} + \frac{k}{m} x = 0 \qquad (2)$$

The quantity $\omega_n$ (the natural frequency in radians per second) can be substituted for $\sqrt{\frac{k}{m}}$ and we obtain

$$\ddot{x} + \omega_n^2 x = 0 \qquad (3)$$

Later uses will require the frequency f to be expressed in Hz (cycles per second) and this is obtained from

$$f_n = \frac{1}{2\pi} \omega_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} = \frac{1}{2\pi} \sqrt{\frac{kg}{W}},$$

where W is the weight of the test mass and g the acceleration of gravity.

The solution of this homogeneous second order differential equation (3) for an undamped simple harmonic motion system is

$$x = X \sin (\omega_n t + \phi) \qquad (4)$$

where $\phi$ is the phase angle between the forcing and response functions. It is evident from this equation that the displacement of the mass, x, is a function of the displacement of the forcing function X, the natural frequency of the system, $\omega_n$, and the time t.

All systems contain damping to a certain extent. This is evident from the fact that nothing will continue in motion indefinitely. The damping force is associated with the dissipation of energy, usually in the form of heat, and may be a function of many different variables (displacement, velocity, stress, etc). However, for the sake of simplicity, a type of damping (viscous damping), which is a function of the velocity $\dot{x}$, is frequently used and will so be used here. Also, a proportionality constant c appears in calculations of the viscous damping.

Unless an item is completely stationary (or coming to rest after the removal of some force), there is present a forcing function (external from the system) to excite the system. In our test application, which uses a hydraulic vibrator, the force produced is harmonic and is equal to $F_o \sin \omega t$. Including these quantities in the equation of motion (2) we obtain

$$m\ddot{x} + c\dot{x} + kx = F_o \sin \omega t \qquad (5)$$

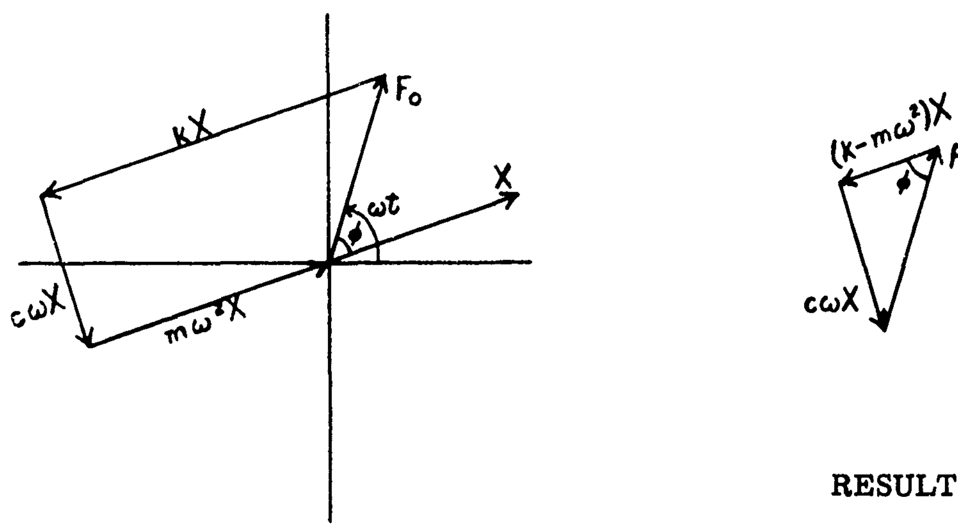
If the solution for x, stated earlier, is substituted into this equation, the result is

$$m\,\omega^2 X \sin(\omega t - \phi) - c\,\omega X \sin(\omega t - \phi + \pi/2) - k X \sin(\omega t - \phi) + F_o \sin \omega t = 0 \tag{6}$$

Stating this in words we have

Inertia Force + Damping Force + Spring Force + Impressed Force = 0

A vectorial representation of the quantities present in the equation and their relationship is:




RESULTANT

This shows that:

1. The displacement lags the impressed force by the angle $\phi$, which can vary between 0 and 180 degrees.

2. The spring force is always opposite in direction to the displacement.

3. The damping force lags the displacement by 90 degrees and hence is opposite in direction to the velocity.

4. The inertia force is in phase with the displacement and opposite in direction to the acceleration.

The solution for X of this last equation and vectorial representation is:

$$X = \frac{F_o}{\sqrt{(k - m\,\omega^2)^2 + (c\,\omega)^2}} \qquad (7)$$

To get a nondimensional form we can divide by k:

$$X = \frac{F_o/k}{\sqrt{(1 - \frac{m\,\omega^2}{k}) + (\frac{c\,\omega}{k})^2}} \qquad (8)$$

A more useful and simplified form of this is obtained by introducing a few more equalities:

$\omega_n = \sqrt{\frac{k}{m}}$ = natural angular frequency

$\zeta = \frac{c}{c_c}$ = damping factor

$c_c = 2m\omega_n$ = critical damping coefficient

$X_o = \frac{F_o}{k}$ = zero frequency deflection under the action of a steady force $F_o$.

The finalized representation can now be stated as:

$$\frac{X}{X_o} = \frac{1}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]^2}} \qquad (9)$$

The term $\frac{X}{X_o}$ is one of the most important quantities encountered in vibration. It is called the transmissibility and represents the factor by which the zero frequency deflection $X_o$ must be multiplied to determine the displacement amplitude X of the isolated item. $X_o$ can be equated to the amplitude of the input force to yield a solution determining the magnification factor for a system at a particular frequency. It is evident from the last equation that the peak transmissibility Q is a function of the frequency ratio $\frac{\omega}{\omega_n}$ and the damping factor $\zeta$. This relationship is plotted in Figure 2.

The frequency of greatest interest in vibration is the natural angular frequency $\omega_n$ (in rad/sec). This is the point at which resonance occurs, and consequently the ratio $\frac{\omega}{\omega_n} = 1$. Substituting this into (9) we get:

$$\frac{X}{X_o} = \frac{1}{2\zeta} = \frac{1}{2\ c/c_c} \qquad (10)$$

The maximum transmissibility (at $\omega = \omega_n$) is therefore a function of the damping alone. (Of course, the critical damping coefficient, $c_c$, is, in turn, a function of the mass and natural frequency).

The dissipation of energy as heat in damping has another effect on the transmissibility and resonant point: Because of the particular configuration of various mounting systems, the heat does not escape into the surroundings but causes shear mounts themselves to become hot and to soften. As that occurs, the damping decreases and the resonant frequency shifts lower. This, in turn, causes the transmissibility to increase slightly.

## DISCUSSION OF TEST APPARATUS AND PROCEDURES

The test fixtures and associated hardware to conduct studies on shear mounts were developed at Picatinny Arsenal. Both the static load deflection and vibration fixtures simulate a single-degree-of-freedom system and were originally designed to test mounts according to military specifications, including Picatinny Arsenal Specification PA-PD-2522. This equipment is shown in Figures 1a, 1b, and 3.

With the existing mounting hole pattern in the end plates of both fixtures, two different size mounts can be tested: one with 3 3/4 inches between the centers of four holes arranged in a square, and one with 5 3/4 inches between the centers. Two mounts are required, and for vibration a mass simulating the item to be protected is suspended between them. In the vibration fixture the weight of the suspended mass can be varied from 174 pounds to 610 pounds, depending on the natural frequency of the system that is desired. After establishing the value for dynamic spring rate and choosing a value for either the natural frequency or weight, the remaining variable (weight or frequency) can be determined from the formula

$$f_n = \frac{1}{2\pi}\sqrt{\frac{kg}{W}}$$

Each butadiene styrene mount used had a static spring rate of 470 pounds per inch and a dynamic spring rate, as supplied by the manufacturer, of 610 pounds per inch. (These values are doubled for calculations, since the mounts are used in pairs). Experience has shown that most shipping containers have a suspension system with a natural frequency between 6 and 9 Hz. Therefore, it was decided to use a suspended mass of 190 pounds for the pair of mounts so that a natural frequency in vibration of approximately 7.9 Hz would result.

To perform the static load deflection tests, the static load deflection fixture (similar to the one used in vibration) and a compression test machine were used. Figure 4 shows this configuration. A ram is fastened to the two mounts which are in turn connected to a rigid outer structure. After setting the fixture in the compression machine, each pair was cycled (flexed) twice and then deflected to a total length of three inches. All deflections were run at a speed of one inch per minute.

In order to follow as closely as possible existing military specifications (PA-PD-2522) each pair was cycled twice to 2/3 of its maximum deflection (3 inches). Consequently, the two flexes consisted of two inches total deflection, followed by the recorded and calculated static load deflection to three inches. It is true that the compression table is moving and therefore the test not truly static, but the excursion is slow enough to fall in the area of static loading.

From past experience it appeared that the temperature was a main factor in the failure of butadiene styrene mounts. It is generally recognized that when temperatures of approximately 200$^{o}$F are reached, degradation of this material occurs. Even when vibrated at the temperature extreme of -65$^{o}$F or at ambient room temperature, the heat build-up in the mount itself is very rapid. In the course of our ambient temperature vibration a high temperature (in the vicinity of 200$^{o}$F) was reached, as was evident from smoking and temperature measurements. Due to this factor, the limited supply of mounts, and convenience, the butadiene styrene mounts were vibrated only at ambient temperature.

An electro-hydraulic vibrator with a total force capability of 30,000 pounds was used in this phase of the study. A pictorial view is shown in Figure 7. The rigid outer structure of the shock and vibration fixture (it is also used in shock mitigation research) was bolted firmly to the head of the vibrator. The two mounts, with the suspended 190 pound mass[1] between them, then were slid into place and secured. An accelerometer was attached to the vibrator head and one to the suspended mass so that the input and response could be recorded and the transmissibility calculated.

To find the transmissibility and resonant frequency, an input of 0.125 inches DA was used and the assembly was vibrated from 2 to 10 Hz at 1 octave per minute. A typical curve of this procedure is shown in Figure 8. All tests were run in the shear plane (normal usage) of the mounts. Different resonant points and transmissibilities would be encountered in the compression plane.

The vibrations at $f_n$ were performed with inputs of 0.125, 0.2, 0.25, and 0.5 inch DA (double amplitude). (Three pairs were vibrated at 0.125, 0.25, and 0.5 inch each, and two pairs at 0.2 inch.) These values were chosen to provide as braod a time range as possible.

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[1]The terms mass and weight are, of course, not equal, but according to common practice they are used interchangeably here

The times to failure were recored and a curve plotted. This procedure produced an S-N type curve (strain vs number of cycles to failure).

As a shear mount becomes hot due to vibration, $f_n$ shifts slightly lower, and this causes a problem in maintaining the input frequency at the resonant point. MIL-STD-810 requires a continuous retuning (chasing) to resonance and PA-PD-2522 requires retuning at the beginning of each period for the resonance dwell test. Because this vibration was to be continuous until failure occured, and due to the rapidity with which $f_n$ changed, it was decided to chase the shifting frequency uninterruptedly until $f_n$ stabilized.

Since temperature is presumed to play an integral part in the failure of these shear mounts, an effort was made to record it during vibration. Due tc the intrinsic properties of butadiene styrene, no available adhesives will adhere to its surface. Therefore a tiny thermocouple was placed between the plate of the mount and one of the end plates of the fixture. In every case, however, the fragile wires broke as a result of the vibration. An attempt also was made to secure a sensor to the surface of the mount by wrapping tape around its circumference several times. Here again failure was encountered, since the tape was broken by the flexing of the mount. However, by attaching a probe to the surface after vibration had been stopped, a reading of approximately 180°F was found. Since the mounts were already beginning to cool, the temperature during fatigue and failure must have been somewhat above this value.

As was stated earlier, after stabilization no appreciable change in Q or $f_n$ was noted as failure approached. This would give the impression that failure had not occurred at all. Indeed, if physical signs of deterioration had not been observed visually, it would not have been detected at all. Therefore, it was decided to evaluate the extent of loss of static spring rate in the mounts that had been fatigued to failure. The spring rate had been 470 pounds per inch before vibration.

Five pairs were chosen at random (sets 2, 4, 5, 6, and 8) and again subjected to the load deflection test as done earlier. A comparison at excursions of 1, 2, and 3 inches for all those deflected after failure was made.

## DISCUSSION O' TEST RESULTS

The objectives of this study had been to develop a method of testing the fatigue characteristics of shear mounts, define when and why failure occurs, and begin the testing of selected samples. To this end, a method was established using vibration at resonance to varying inputs; failure was investigated and defined during the vibration.

When the mounts composed of butadiene styrene were received from the manufacturer, they were visually inspected for observable defects which could be a factor in poor performance or premature failure. The load deflection tests, subsequently imposed on each pair, yielded values that conformed, within the required 5%, with the manufacturer's listed static spring rate of 470 pounds per inch (940 lb/inch for a pair). The correlation among the 12 pairs also was within 7.25%. A sample curve for the load deflection test is given in Figure 5, and a tabulation of all the load values at deflections of 1, 2, and 3 inches in Table 1. Also a graph of all the points (indicating the scatter) can be found in Figure 6.

Each mount (or, as is more appropriate in this case, pair) has its own inherent characteristics even though they are all supposedly molded alike. These characteristics (specifically transmissibility and resonant frequency) may in some cases vary considerably from one to another; e.g., values for $f_n$ ranged from 8.4 to 9.2 Hz, while the corresponding Q's ranged from 3.1 to 4.4. However, as shown in Table 1, the deflections recorded during the static testing and time durations recorded for the vibration to destruction show no significant correlation to either high or low values of transmissibility or resonant frequency.

The transmissibility test (Fig 8) included vibration from 2 to 10 Hz at 0.125 in DA. At 5 Hz the electro-hydraulic vibrator produces an input acceleration of 0.17 G's and at 10 Hz it is 0.7G's. The values recorded for the suspended mass were always equal to or greater than the input force (below $f_n$) until a peak was reached of between 1.6 and 2.2 G's, depending on the pair under study. All these values are for an input of 0.125 DA, which also was the lowest displacement used in the fatigue tests. At the input of 0.5 DA (largest value used) the resulting acceleration on the mass for the greatest Q reached almost 7 G's.

It was stated above that no correlation exists between the length of time to failure and transmissibility. This statement must, however, be qualified to the extent that a tendency appeared toward greater times to failure at the lower Q values (as might be expected); but not enough tests were performed to substantiate a definite statement. At the value of 0.25 in. DA, three pairs were vibrated and the durations were 12, 11.5, and 11 min, with the corresponding Q values of 3.3, 4.0, and 4.4. The three pairs tested at the input of 0.125 in. DA produced times of 4.5, 4.33, and 3.67 hours and respective Q's of 3.6, 3.4, and 3.9. These last readings do not conform exactly to the pattern; but it will be noted that there is little variation between 3.4 and 3.6 or between 4.5 and 4.33 hours. Vibration imposed on two pairs at 0.2 in. DA produced durations of 23 and 22.5 min for transmissibilities of 3.5 and 3.1. Again there is a minor discrepency; but only two values are available for comparison.

The times to failure at Q's recorded (6.5, 5, and 3 min, and 3.6, 3.3, and 3.3) for the three pairs at the input of 0.5 DA show little correlation. A great deal of difficulty was encountered at this level because the displacements of the mass were great and the failure times extremely short. An error of only a small amount in detecting the exact time of failure seems magnified by the comparable short interval of the durations involved.

Twelve pairs of mounts had originally been received and all twelve were vibrated. However, one pair vibrated at 0.125 DA was thought to have failed, and the test was stopped prematurely. As a result, only two pairs were available for vibration at 0.2 in. A small tear was noted on the surface of one of these mounts and it was removed from the fixture and dissected. Only at this point was it discovered that failure had not actually occurred and the tear was only a surface blemish. Other pairs that had failed also were sectioned (Fig 11) and the extent of the deterioration examined. In many cases it extended through the wall of the mount into the core.

The retuning required during fatigue vibration amounted to approximately 2 Hz. It was a downward shift and the retuning was needed for only a short time at the beginning of each vibration. It became apparent that as the mounts heated up and reached a relatively constant temperature, the frequency became stable. Vibration then proceeded until failure occurred at the new, lower resonant frequency.

Also, a small upward shift in the transmissibility was noted to occur in conjunction with the frequency shift. This apparently was due to the softening of the mounts from the heat as the vibration progressed. In several cases, as the mounts began to fatigue, a slight eccentricity appeared in the vertical motion of the mass. The motion became progressively worse until failure occurred; it was attributed to the failure of one of the pair before the other. In all cases, the mount causing this erratic motion (and the one that became the hottest) did fail first.

Investigation of the fatigue life of the mounts under vibration was the major objective of the program, and the resulting S-N type curve (input vs time to failure) proved very satisfactory. As is evident from Figure 9, the curve becomes asymptotic at small inputs. This was predictable from past experience with elastomeric shear mounts; but the very short life at the large inputs was unexpected. As indicated above, failure proved difficult to establish accurately at the higher inputs, and this might have led to some error in the recorded intervals.

The problems encountered in determining the time to failure were preceded by those of defining what constitutes failure (and what might be its cause or causes). The exact causes are still not definitely known, but from the information obtained a number of good possibilities can be advanced with confidence. For some time heat has been associated with failure. It may actually cause a deterioration of the material (or bond) itself, or combine with the motion of the mounts to exert a softening effect and therefore greater flexing. In either case, there is a definite strain-temperature relationship associated with the material breakdown. Undoubtedly, other factors combine with the motion and heat to contribute to failure.

As to a definition of failure, many considerations had to be explored. One would expect a sharp increase in the transmissibility as failure approaches, but this, as stated above, is not the case. A slight increase, along with a downward shift in natural frequency, appeared early in the fatigue of each pair, but no definite changes were clearly evident at failure. Also, the eccentricity of motion of the mass caused by the unequal softening of the mounts continued for some time before the tests were halted. Consequently, neither could be considered a parameter for failure. At the inputs of 0.125 and 0.2 in. DA, where the durations approached a reasonable level, smoking of

one or both of the mounts appeared during vibration. In more than one instance, the smoke was observed long before failure occurred.

Taking all this and past experience into consideration, failure had to be defined as a visual deterioration (tear in the surface or separation of the bond between the styrene and metal plates) of a mount. Physical defects in a shear mount have, of course, been a frequent cause for rejection. This is one of the requirements of initial inspection and acceptance. Therefore, physical, visual degradation of a mount must be accepted as an adequate parameter of failure.

In an attempt to determine what effect failure (as previously defined) would have on the load deflection characteristics of the mountings, this test was repeated after the completion of fatigue testing. Surprisingly, the load deflection values only showed losses averaging less than 10% from the original values. A sample curve of an earlier and later deflection is shown in Figure 10, and a comparison of the five pairs retested is given in Table 1. One can assume, then, that a large percentage of the isolation capability of the mountings remained despite obvious elastomeric tears or separations. In an emergency situation, therefore, ordnance material utilizing butadiene styrene mounting systems could be shipped despite indications of mount failure.

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4. MIL-M-50508 (revision of PA-PD-2522)

5. MIL-STD-810B

TABLE 1

Vibration fatigue testing at resonance frequency of butadiene styrene elastomeric shear mounts

| MOUNTS (PAIR) NO. | STATIC LOAD (LB) VS DEFLECTION (IN.) 1 in. | 2 in. | 3 in. | MAXIMUM TRANS-MISSI-BILITY Q | NATURAL FREQ fn (HZ) | VIBRATION FATIGUE FAILURE INPUT DA (INCHES) | TIME (MIN) | STATIC LOAD (LB) VS DEFLECTION (IN.) AFTER VIBRATION 1 in. | 2 in. | 3 in. |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 960 | 1830 | 2630 | 3.6 | 8.4 | 0.5 | 6.5 | | | |
| 2 | 930 | 1800 | 2700 | 3.3 | 8.8 | 0.5 | 5.0 | 900 | 1680 | 2610 |
| 3 | 990 | 1890 | 2820 | 3.3 | 8.7 | 0.5 | 3.0 | | | |
| 4 | 945 | 1815 | 2700 | 3.3 | 9.2 | 0.25 | 12.0 | 930 | 1650 | 2445 |
| 5 | 960 | 1800 | 2700 | 4.4 | 8.5 | 0.25 | 11.0 | 915 | 1620 | 2460 |
| 6 | 990 | 1890 | 2790 | 4.0 | 8.7 | 0.25 | 11.5 | 930 | 1635 | 2460 |
| 7 | 960 | 1860 | 2730 | 3.6 | 8.7 | NO DATA | | | | |
| 8 | 990 | 1875 | 2745 | 3.4 | 8.7 | 0.125 | 260 | 915 | 1605 | 2445 |
| 9 | 975 | 1875 | 2775 | 3.6 | 8.8 | 0.125 | 275 | | | |
| 10 | 945 | 1815 | 2670 | 3.9 | 9.0 | 0.125 | 220 | | | |
| 11 | 930 | 1800 | 2660 | 3.5 | 9.0 | 0.2 | 23 | | | |
| 12 | 975 | 1875 | 2760 | 3.1 | 8.8 | 0.2 | 22 | | | |

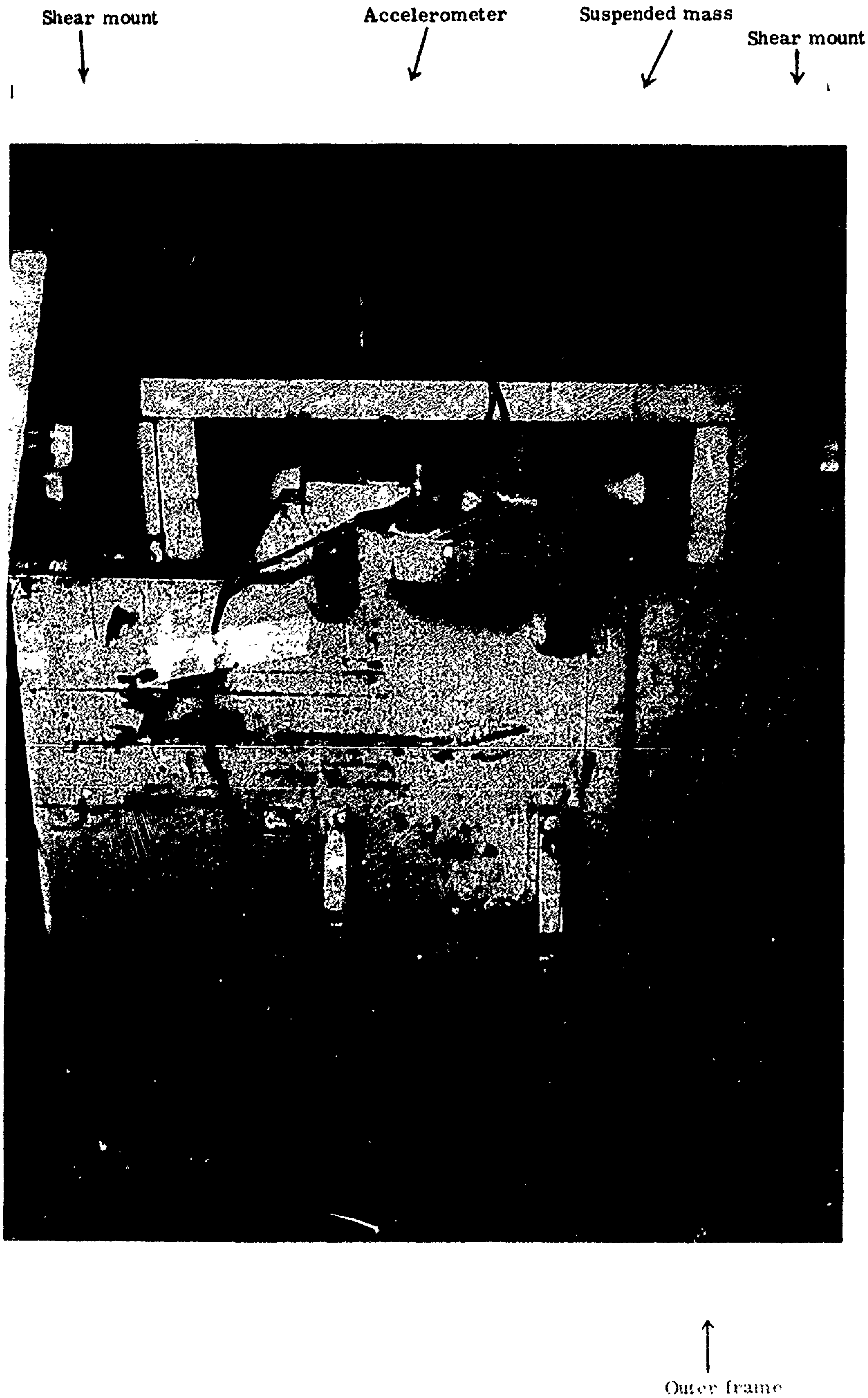



Fig 1a Vibration test assembly

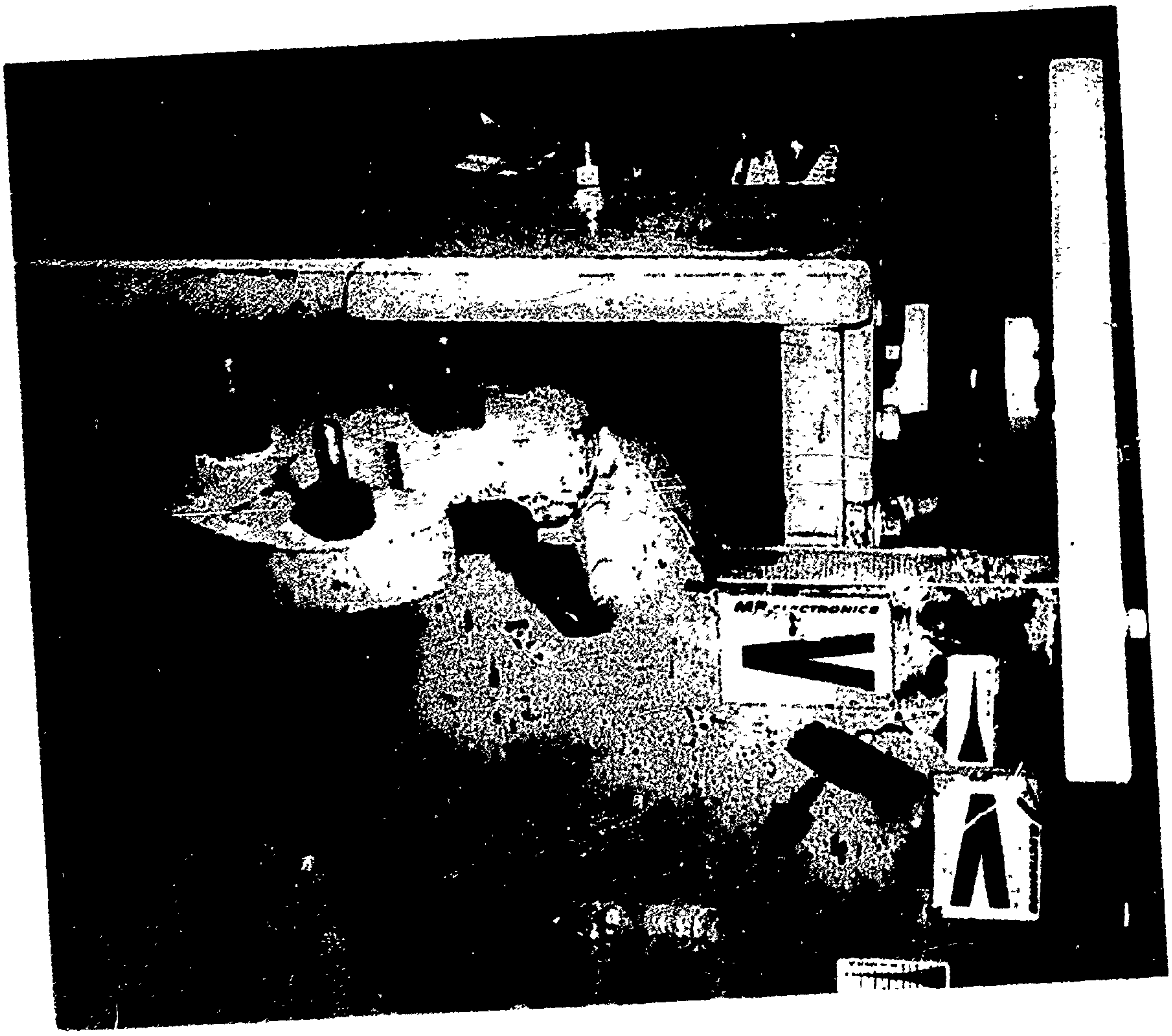



Fig 1b Vibration test assembly (view of shear mount)

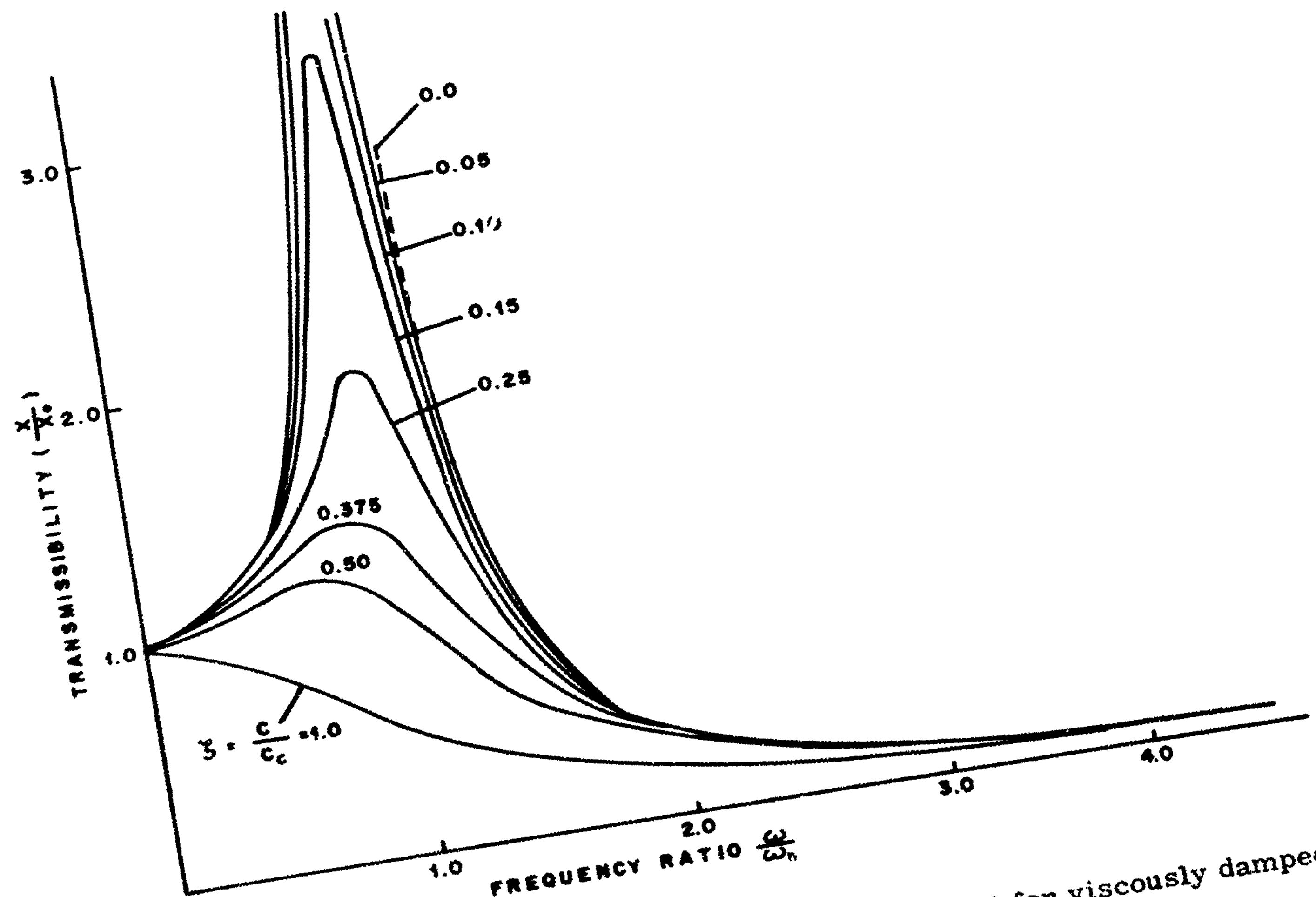



Fig 2  Vibration response ratio (transmissibility vs frequency) for viscously damped system

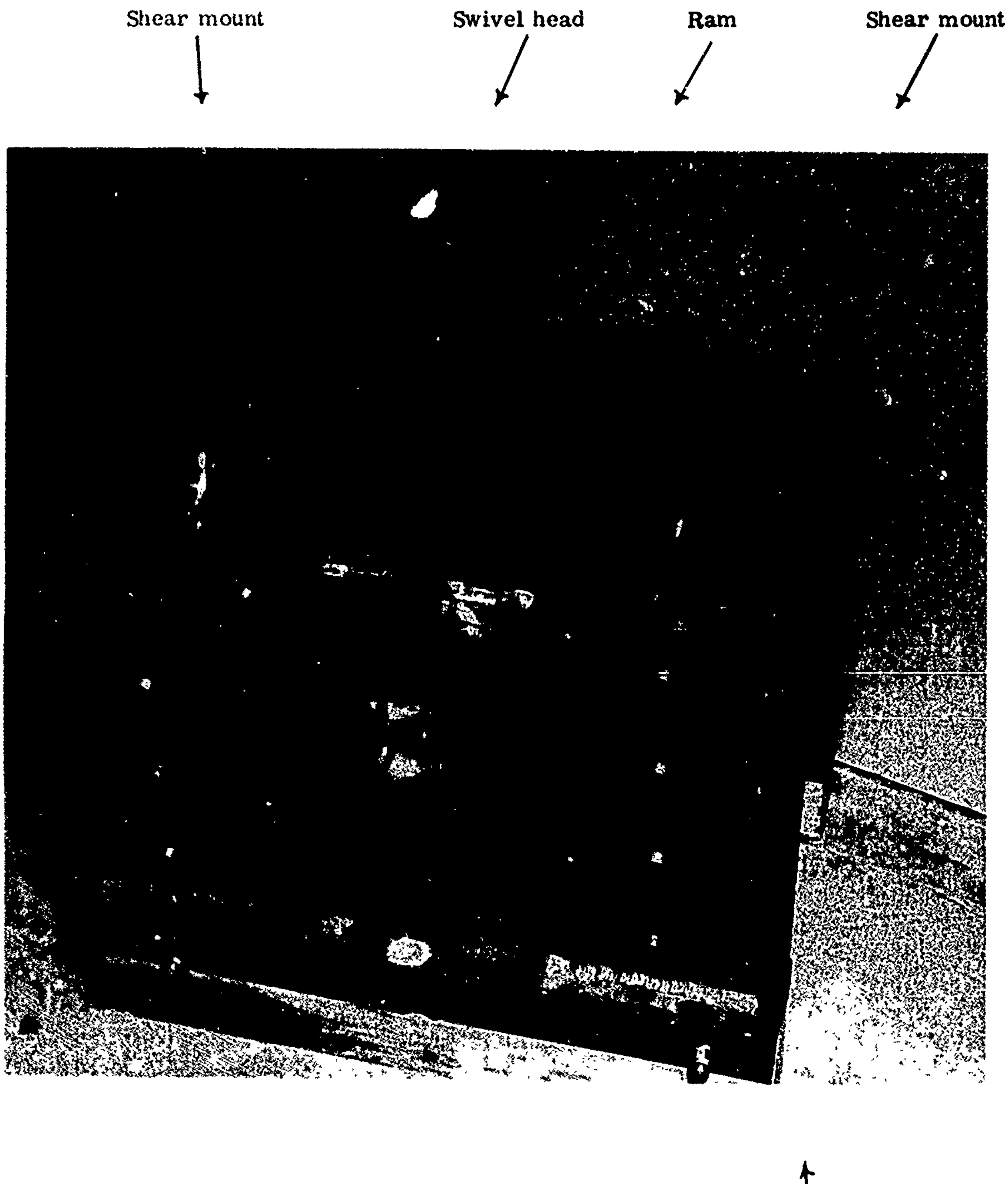



Fig 3 Static load deflection test assembly

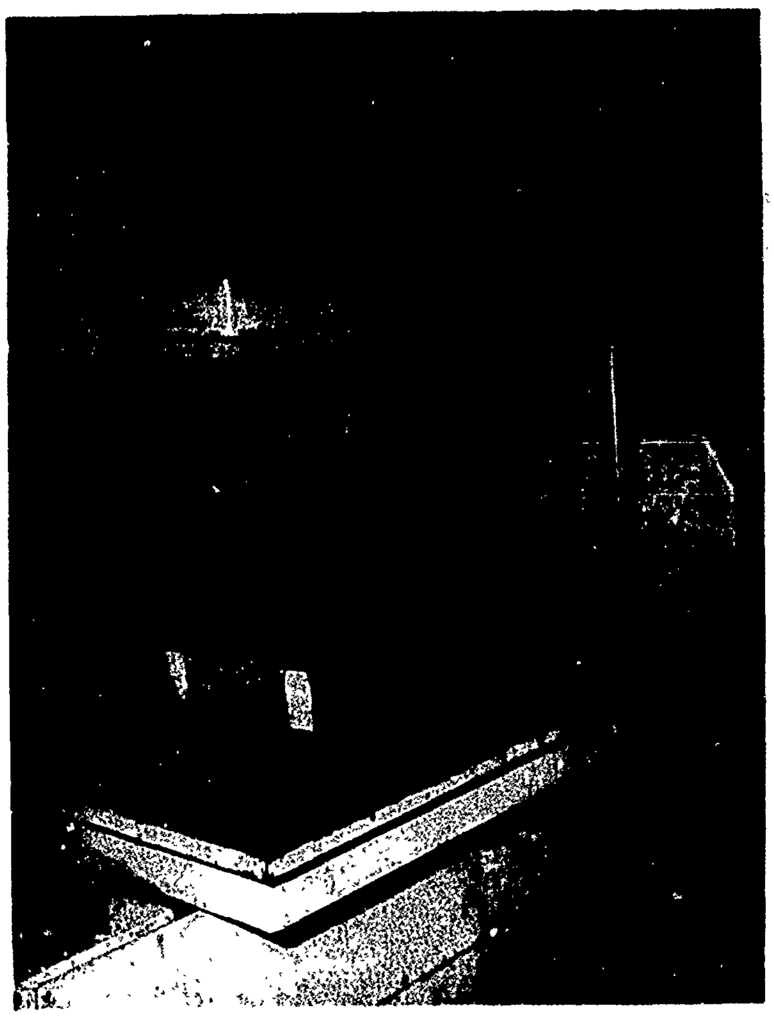



Fig 4 Load deflection test assembly in compression test machine

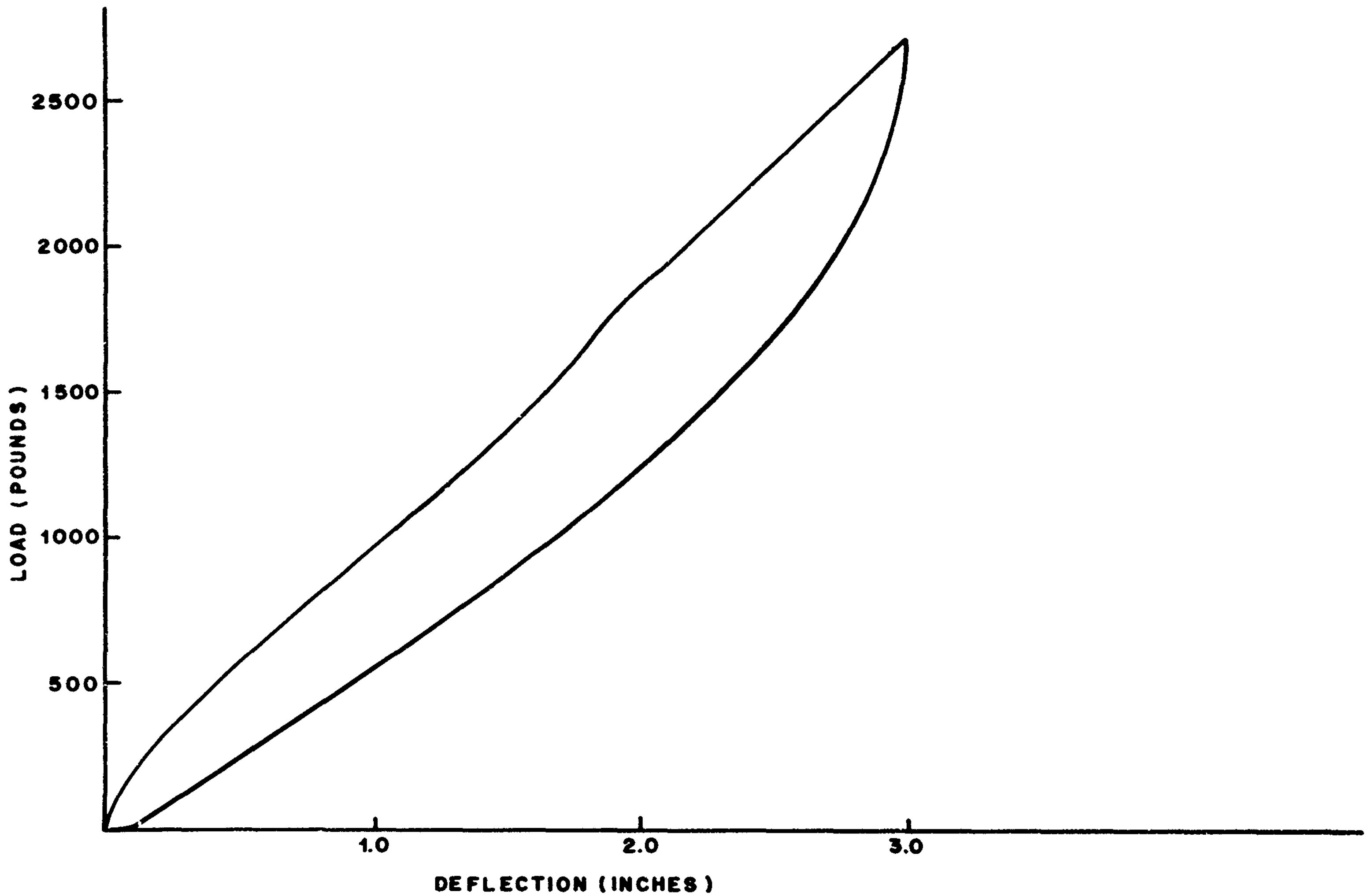



Fig 5 Sample curve of static load deflection test

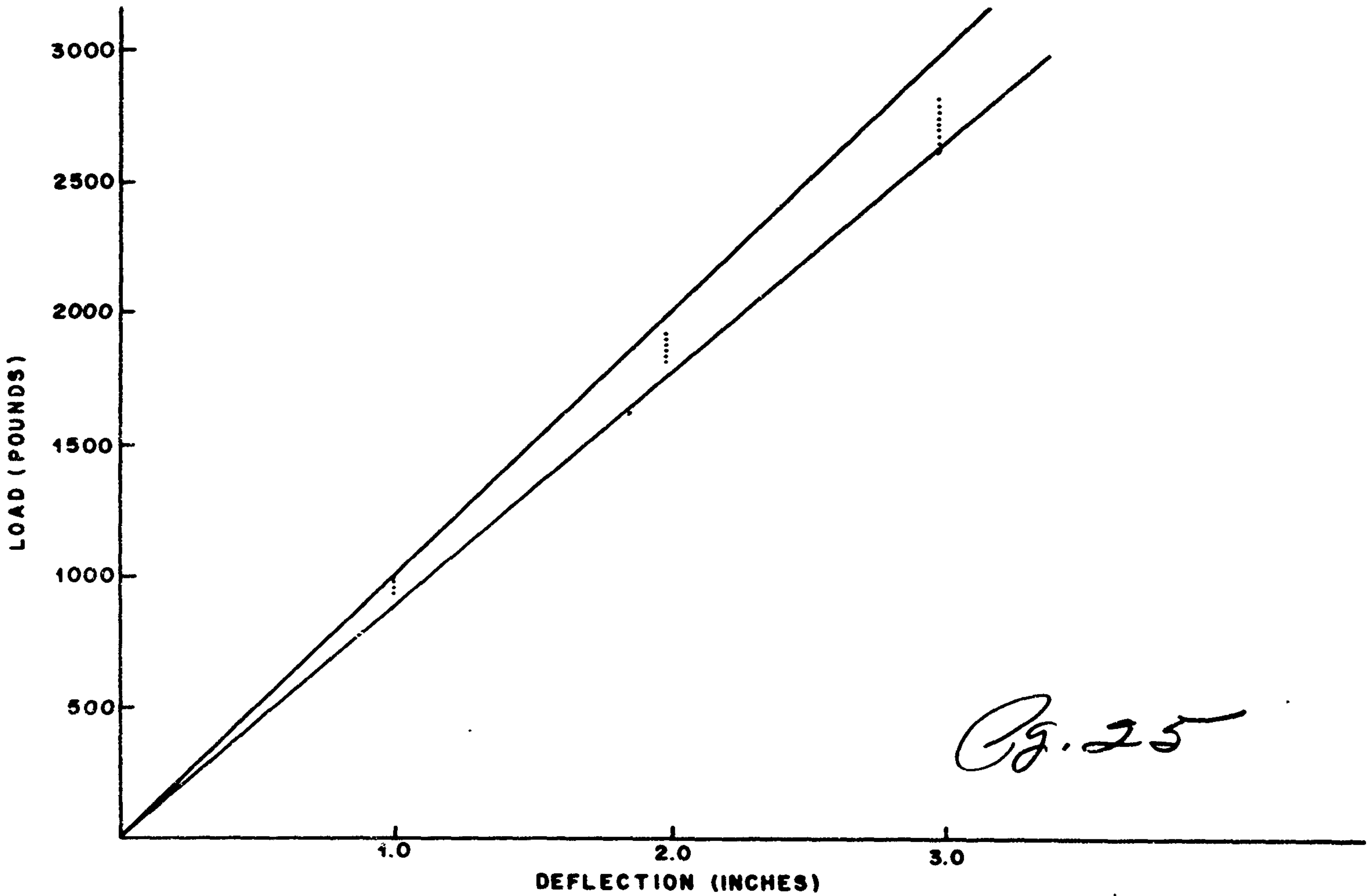
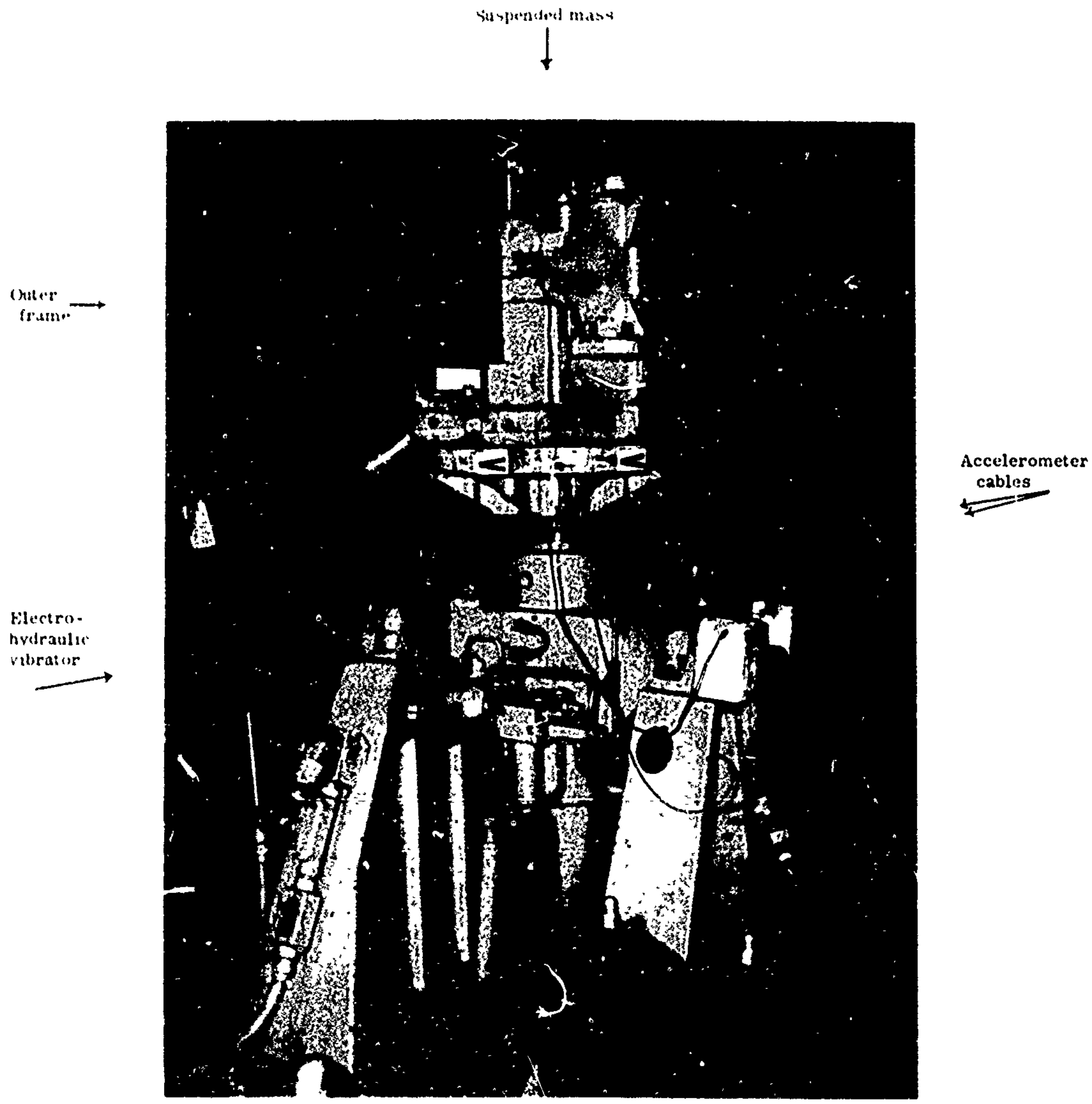



Fig 6 Scatter of static load deflections
(pounds vs inches)




Fib 7 Vibration test fixture mounted on hydraulic vibrator

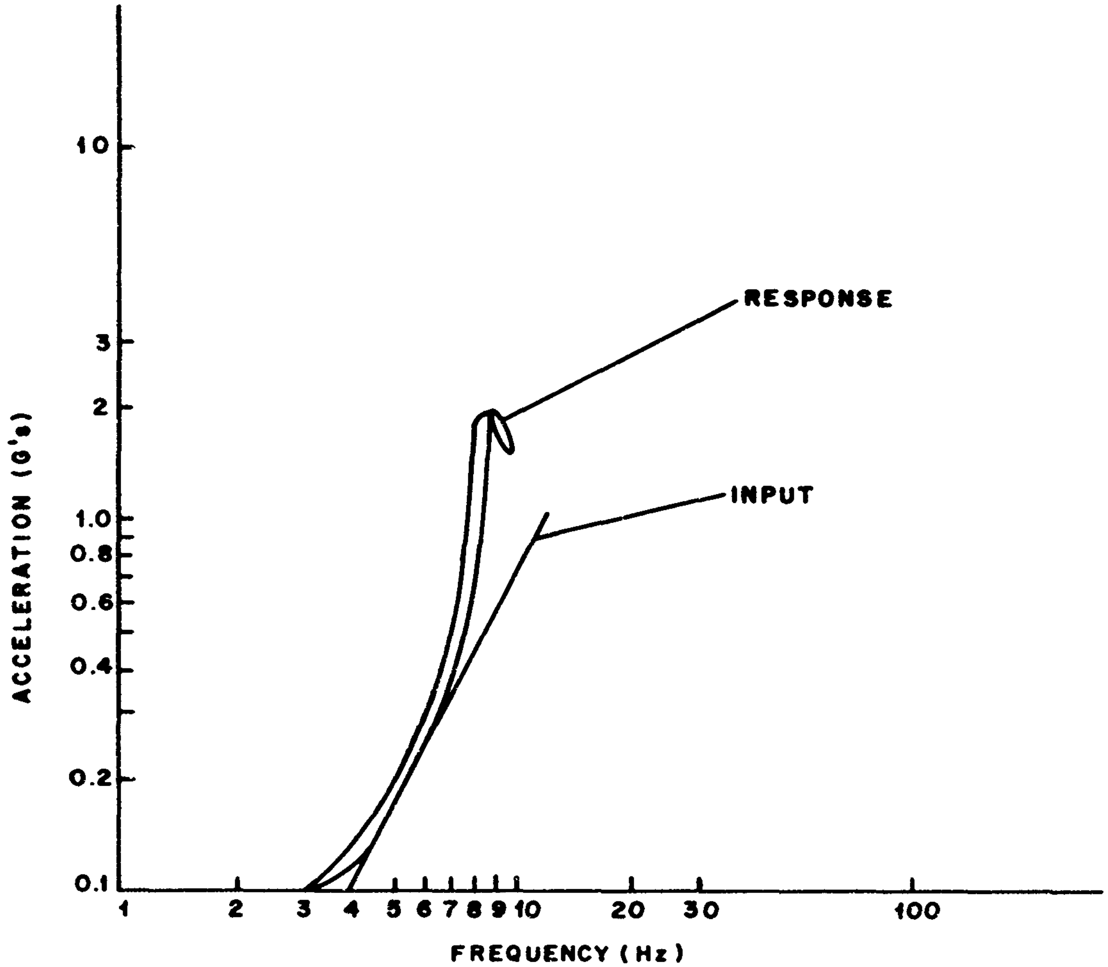



Fig 8 Transmissibility curve

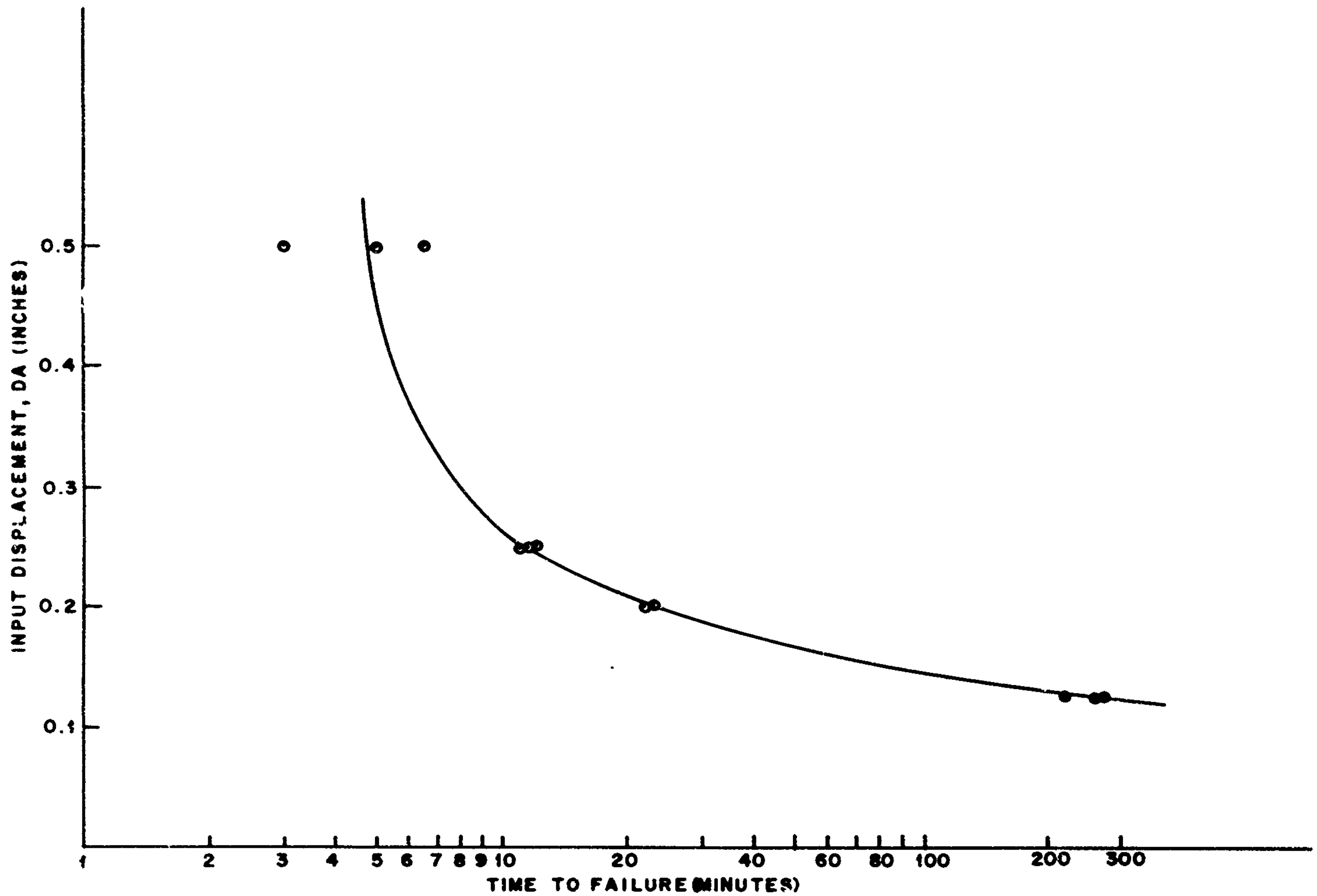



Fig 9 Vibration fatigue failure

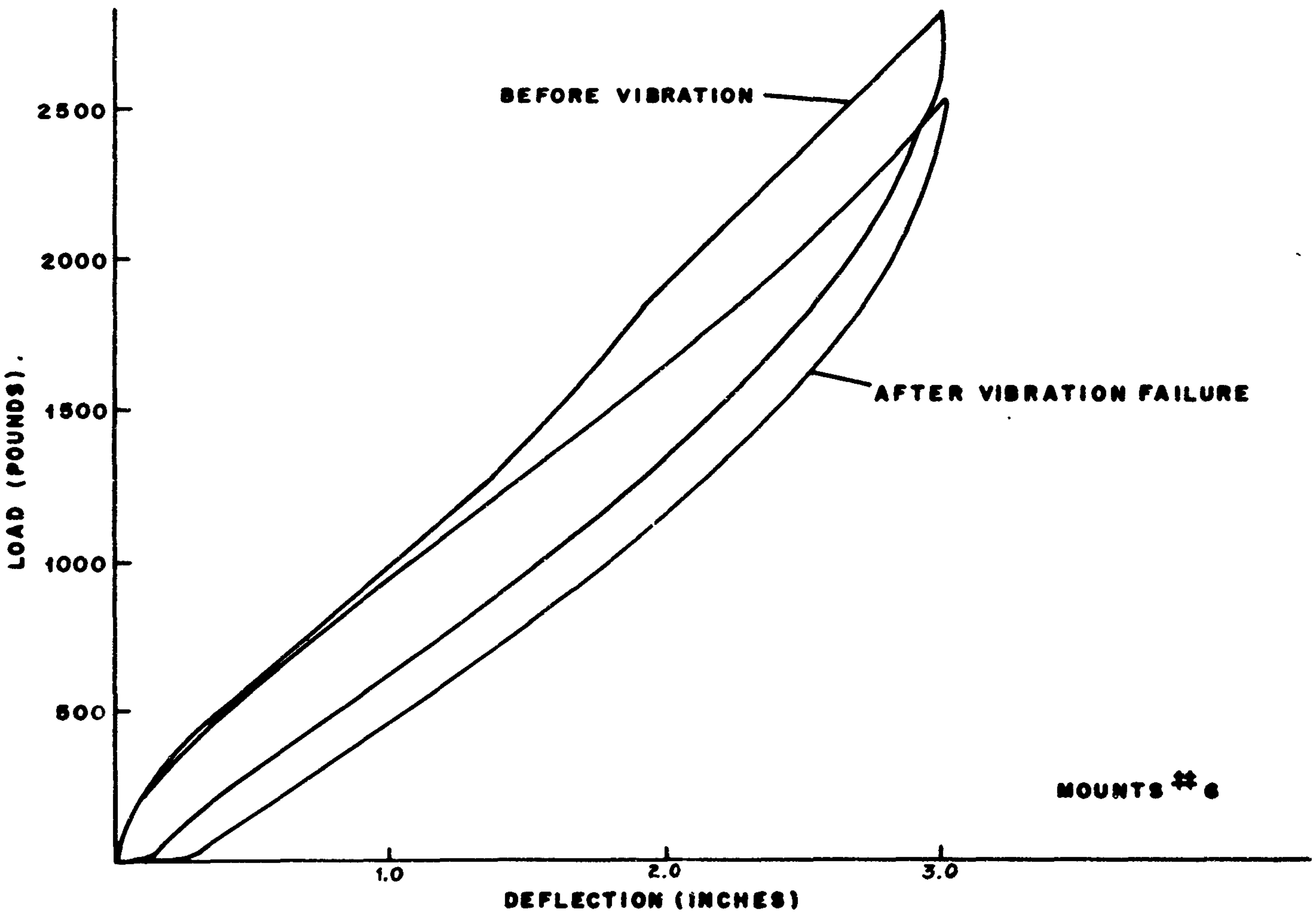



Fig 10 Comparison of static load deflections
(before vibration and after failure)

Fig 11 View of tear (failure) in shear mounts

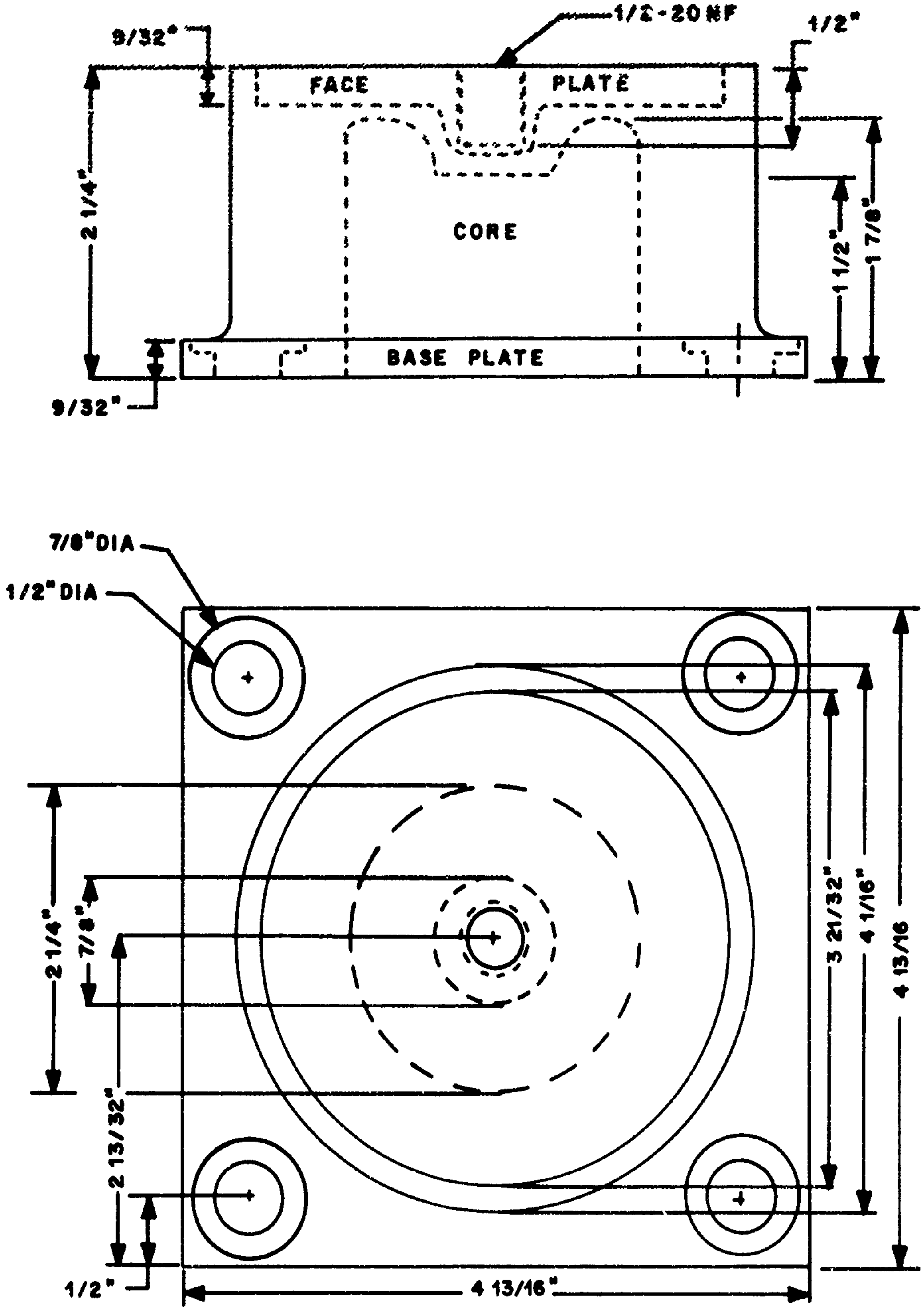



Fig 12 Physical configuration of butadiene styrene mounts subjected to fatigue

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13. ABSTRACT A method of measuring the fatigue life of elastomeric shear mounts was developed. It included vibration testing at resonance to failure at different inputs, and plotting the time to failure vs input displacement for each set of mounts tested.

Twelve pairs of mounts composed of butadiene styrene elastomer were subjected to static load deflection and vibration testing. The nominal static spring rate of one mount was 470 pounds/in. The load deflections yielded results within 7% of this value.

During initial vibration the transmissibility (Q) was found to vary between 3.1 and 4.4 and the resonant frequency ($f_n$) between 8.4 and 9.2 Hz. Eleven pairs of mounts were fatigue-vibrated at double amplitudes (DA) of 1/8", 0.2", 1/4" and 1/2". The durations to failure were approximately four hours for 1/8 inch and five minutes for 1/2 inch.

What actually constitutes failure in a shear mount has been a matter of concern for some time. After investigating changes, if any, in frequency, transmissibility, etc, failure was defined as a visual tear or separation of the mount.

Upon completion of the fatigue vibration testing, five pairs of mounts were again subjected to load deflection testing. The loss of static spring rate amounted to less than 15% for all five deflected up to 3 inches, and in most cases it was less than 10%.


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| 14. KEY WORDS | LINK A | | LINK B | | LINK C | |
|---|---|---|---|---|---|---|
| | ROLE | WT | ROLE | WT | ROLE | WT |
| Butadiene styrene elastomer | | | | | | |
| Sandwich-type shear mounts | | | | | | |
| Fatigue testing | | | | | | |
| Double amplitude | | | | | | |
| Resonance to failure | | | | | | |
| Imput levels | | | | | | |
| Load deflection and vibration testing | | | | | | |
| Static spring rates | | | | | | |
| Transmissibility vibrations | | | | | | |
| Damping constant | | | | | | |
| Displacement of exciting force | | | | | | |
| Phase angle | | | | | | |